

The Integration of Liquid Cryogen Cooling and Cryocoolers With Superconducting Electronic Systems

**Michael A. Green
Lawrence Berkeley National Laboratory
University of California
Berkeley CA 94720, USA**

I. ABSTRACT

The need for cryogenic cooling has been a critical issue that has kept superconducting electronic devices from reaching the market place. Even though the performance of many of the superconducting circuits is superior to silicon electronics, the requirement for cryogenic cooling has put the superconducting devices at a serious disadvantage. This report discusses the process of refrigerating superconducting devices with cryogenic liquids and small cryocoolers. Three types of cryocoolers are compared for vibration, efficiency, and reliability. The connection of a cryocooler to the load is discussed. A comparison of using flexible copper straps to carry the heat load and using heat pipe is shown. The type of instrumentation needed for monitoring and controlling the cooling is discussed.

II. INTRODUCTION

The various options for cooling superconducting electronic systems include; 1) boiling liquid cryogen cooling using either liquid helium or liquid nitrogen (For some devices liquid neon may be an option, but liquid neon is expensive to use as a throw away coolant.), 2) cooling with relatively small cryocoolers, or 3) cooling with more conventional liquid helium or liquid nitrogen refrigerators. This report will deal only with the first two options, because most electronics applications for superconductivity have low heat loads and they are housed in a limited space. Superconducting electronic systems that require larger refrigerators will have refrigerators that are individually tailored to their cryogenic heat load.

Connecting the cryogenic cooling to the heat load is often the biggest technical issue when one looks at integrating cooling with superconducting electronics systems. Laboratory electronic systems are usually immersed in a pool of liquid helium (LTS systems) or liquid nitrogen (HTS systems). Heat transfer from a device immersed in boiling cryogenic coolant is rarely an issue. Device vibration is not an important issue for a cryogen bath-cooled device that is located in the laboratory

When one uses a cryocooler as the method for cooling the electronic device, a number of issues come into play. They are; 1) the vibration characteristics of the cryocooler and the method used to isolate the cooler from the device, 2) The temperature difference between the electronic device and the cryocooler cold head, 3) the reliability of the cryocooler, and 4) the cost of purchasing and running the cryocooler. One of the approaches that have been used for connecting cryogenic devices to a cryocooler is to use a liquid cryogen interface. If this is not done correctly, the results can be bad. This report shows a method for isolating the vibrating cold head from the device being cooled while transferring the heat from the electronic load to the cold head using a boiling and condensing liquid cryogen.

III. COOLING WITH LIQUID CRYOGENS

Table 1 presents various parameters for cryogenic fluids that might be used to cool superconducting electronics components. The fluids shown in Table 1 include helium, hydrogen, neon, and nitrogen. Only liquid helium or liquid nitrogen would normally be used to cool an electronic component directly, but the other two fluids may be used as an interface fluid between the cold head of cryocooler and the electronic component being cooled.

TABLE 1. Parameters of Cryogenic Fluids [1]

Fluid Parameter	Cryogenic Fluid			
	He	H ₂	Ne	N ₂
Triple Point Temperature (K)	2.17	13.8	24.6	63.1
Triple Point Pressure (atm)	0.051	0.070	0.423	0.128
Boiling Temperature at 1 atm T _b (K)	4.22	20.4	27.2	77.3
Liquid Density at T _b (kg m ⁻³)	125	70.8	1212	808
Critical Temperature (K)	5.19	32.3	44.4	126.1
Critical Pressure (atm)	2.21	12.92	27.1	33.8
Heat of Vaporization (J kg ⁻¹)	20900	442000	86000	199700
C _p Saturated Liquid (J kg ⁻¹ K ⁻¹)	~2500	~9800	~440	~2040
C _p Saturated Gas (J kg ⁻¹ K ⁻¹)	5200	14200	1030	1040
Enthalpy Triple point to 300 K (J kg ⁻¹)	~1578000	~4400000	~367000	~432000
Thermal Cond. Liquid at T _b (W m ⁻¹ K ⁻¹)	0.027	0.119	~0.04	0.140
Thermal Cond. Gas at T _b (W m ⁻¹ K ⁻¹)	0.011	0.021	0.014	~0.0075
Viscosity Liquid at T _b (kg m ⁻¹ s ⁻¹)	3.53x10 ⁻⁶	1.34x10 ⁻⁵	~7x10 ⁻⁵	1.58x10 ⁻⁴
Viscosity Gas at T _b (kg m ⁻¹ s ⁻¹)	0.91x10 ⁻⁶	1.05x10 ⁻⁶	4.32x10 ⁻⁶	~5.3x10 ⁻⁶

When one uses a boiling fluid to cool an electronic device, the operating temperature is close to the normal boiling point at 1 bar. For most applications, one would normally use liquid helium or liquid nitrogen as a coolant. There may be special cases when one would consider using liquid neon, provided the gas is recovered. One can extend the range of operating temperature for a bath-cooled device by using the coolant in the range of pressures from the triple point to 0.8 times the critical point pressure.

On most spacecraft, one would use helium as a liquid cryogen of choice (regardless of the device operating temperature) because the total enthalpy per unit cryogen mass available from the boiling temperature to 300 K is highest for any fluid (except for hydrogen). Hydrogen is a special case. It has good heat transfer properties and the largest total enthalpy of any fluid. The problem is its flammability. Solid hydrogen can be used for space cryogenic cooling of devices (above 13 K) on satellites launched using non-manned expendable rockets. NASA space shuttle safety regulations specifically prohibit the launch of any satellite carrying liquid or solid hydrogen. For very long life a missions at temperatures less than 13 K one must use a combination of solid hydrogen at 13 K and super-fluid helium at 1.7 K. These missions can only be launched on non-manned expendable launch vehicles.

IV. CRYOGENIC COOLERS

Cryogenic refrigerators come in two general forms; 1) open cycle coolers where the working fluid in the cooler comes in direct contact with the object being cooled, and 2) closed cycle coolers where the refrigerator working fluid does not come in contact with the load. When the word cryocooler is used in this report, the author is referring to the second type of cooler [2]. Cryocoolers typically have one or more cold heads that must be connected to the object being cooled.

Open cycle machines are generally designed for larger heat loads than are commonly found in superconducting electronics applications. Open cycle machines permit one to cool electrical leads with gas from the cooling circuit, which allows the heat coming down the leads to be intercepted by the gas stream going up the lead. When the open cycle refrigerant is a gas with a large specific heat (such as helium) one can intercept the heat coming down the leads with a relatively low gas mass flow. For a helium system, this can reduce the refrigeration needed for lead cooling by a factor of five.

A modified cryocooler may have a separate stream from the compressor that connects the cold heads to the load through extra heat exchanges. There usually is a final stage of J-T expansion. In this case, the working fluid in the extra circuit can be in direct contact with the load. The working fluid in the extra circuit may or may not be circulated by the same compressor that is used for the rest of the cryocooler. The modified cryocooler system is usually not as reliable as the cryocooler itself. For that reason, the modified cryocooler is not used for many applications.

Cryocoolers have been built using a variety of refrigeration cycles. Most cryocoolers are of three general types: 1) the most common cryocoolers use the Gifford McMahon (GM) cycle. Single stage GM machines can be used down to 25 to 30 K; two stage machines can be used down to temperatures as low as 2.5 K. GM machines have regeneration and pistons in the cold heads. The moving pistons are a source of vibration. The piston seals are often a cause of machine failure. 2) Pulse tube coolers have recently become commercially available. Single stage pulse tube machines can go down to 25 or 30 K; two stage machines can go down to 2.5 K. Pulse tube machines have no moving parts in the cold head. As a result, pulse tube machines produce much less vibration and should be more reliable than GM machines. 3) Stirling cycle machines are characterized by their high thermodynamic efficiency (in some cases up to three times less input power than a comparable GM machine). The cold head and compressor are often built into a single unit. This is a source of machine inefficiency in many Stirling cycle machines. The Stirling cycle machine cold head contains both moving pistons and regeneration materials. Because of the moving pistons, Stirling units are usually not vibration free.

At least one manufacturer produces a two-piston Stirling machine with linear-drive compressors in a single head. By electronically controlling the phase of the motion of the two pistons, vibration levels for this machine are greatly reduced. On the face of it, Stirling machines should have the same inherent reliability as a GM machine, but there are manufacturers who claim that their machines are substantially more reliable than the typical GM machine. The machines, for which this claim is made, are machines that have been specifically designed for space cooling applications. In recent years, steps have been taken to increase the reliability of all types of cryocoolers. As time goes on one can expect more reliable machines to become commercially available.

Cooler orientation with respect to gravity may be an issue (particularly with the pulse tube coolers). One should check with the cooler manufacturer to see what the effect of orientation of the cold head has on the refrigeration output for the cooler of interest.

V. CONNECTION OF THE COOLER TO THE LOAD

One of the biggest problems in cooling superconducting electronics devices is connecting the cold head of the cooler to the device being cooled. Many electronic applications require very low vibration. Since electronic devices are usually low-mass devices, they are subject to the same level of acceleration seen by the cold head of the cooler, when the device is mounted on the machine cold head.

Vibration isolation can be a key element in the integration of the cooler with the electronic device. In general, the device should be mounted onto a stiff substrate that conducts heat well. This substrate should be connected to room temperature through stiff cold mass supports. Vibration isolation from the cold head can be accomplished by flexible connections from the cooling head (heads) to the load (loads) being cooled. The problem with vibration isolation using a flexible member is the temperature drop between the electronic component being cooled and the cold head. The temperature drop problem is further complicated by heat leaks down the electronic device leads that go from room temperature (or the first stage temperature of the cooler when a two stage cooler is used) to the electronic component being cooled. The literature has a number of papers that describe the issue of how to intercept heat that is entering a device from a higher temperature through its leads [3],[4].

One approach to vibration isolation is to use flexible connectors (multiple thin wires or straps) fabricated from copper that has a high residual resistance ratio RRR (RRR is the ratio of the electrical resistivity at 273 K to the electrical resistivity at 4 K). Copper that has a low electrical resistivity has a high thermal conductivity. At liquid nitrogen temperature, copper with an RRR of 10 or more will have a thermal conductivity of about $800 \text{ W m}^{-1} \text{ K}^{-1}$. At 20 K or 4.2 K, the resistivity will vary greatly depending on the RRR of the copper. For example, RRR = 300 copper will have a thermal conductivity of $5010 \text{ W m}^{-1} \text{ K}^{-1}$ at 20 K and $1880 \text{ W m}^{-1} \text{ K}^{-1}$ at 4.2 K. A RRR = 5 copper will have a thermal conductivity of $144 \text{ W m}^{-1} \text{ K}^{-1}$ at 20 K and $28.8 \text{ W m}^{-1} \text{ K}^{-1}$ at 4.2 K. The temperature drop along a strap is proportional to its length and average thermal conductivity and it is inversely proportional to its cross-section area. In general, the total cross-section area of a flexible strap is much smaller than the area of the cryocooler cold head. For the best vibration isolation the strap must be as long as one can make it. As a result, the temperature drop along the strap may be larger than is desired. The joints that connect the flexible strap to the load and the cryocooler cold head add to the temperature drop. For devices that operate at 4 K and produce large amounts of heat (say $>500 \text{ mW}$) the use of flexible straps to isolate vibration may not be best thing to do. For devices that produce heat in the 1 to 10 mW range, flexible copper straps may be perfectly adequate in all temperature ranges.

An approach for eliminating the effects of heat leaks down the leads into the electronic device is to immerse the device in a cryogenic fluid. Any of the four fluids in Table 1 can be used (including hydrogen, which has very good thermal properties) provided one does cryogenic system design correctly. A device that operates in a cryogen bath can be operated at almost any temperature in the range from just above the triple point to near the critical point for the selected fluid. The vessel containing the load and surrounding cryogenic fluid must be vacuum leak tight and it must be designed for the expected range of operating pressures. A completely closed cryogenic system must be designed to withstand high pressures at room temperature or it will be difficult to control the final temperature of the device being cooled.

The key to connecting the cooler cold head with the load immersed in a cryogenic fluid is creating a condenser that causes the cryogen that the cools the load to circulate [5]. Simply immersing the cold head of the cooler into a bath of liquid cryogen does not work if one wants a low temperature drop

between the source of the heat and the cold head to be low. The difference between making the connection from the cold head to the load correctly and not making the connection correctly is a factor of 10 to 1000 (depending on the case) difference in the effective heat transfer coefficient between the cooled load and the cryocooler cold head. Furthermore, excellent vibration isolation can be achieved because the distance from the load to the cold head can be increased and the elements connecting the two can be flexible within the pressure constraints of the cooling circuit. Figure 1 shows a schematic of a heat pipe system for transmitting heat from a heat producing electronic device to the cold head.

For the four fluids shown in Table 1 one can cool a load over the following temperature ranges; helium from 3.3 to 5.0 K, hydrogen from 15 K to 30 K, neon from 25 K to 40 K and nitrogen from 65 K to 100 K. If one is willing to use oxygen as a cooling fluid, the range of operation can be extended from 55 K to 100 K. The pressure in the cooling circuit will be the saturation pressure associated with a temperature that is close to the cold head temperature.

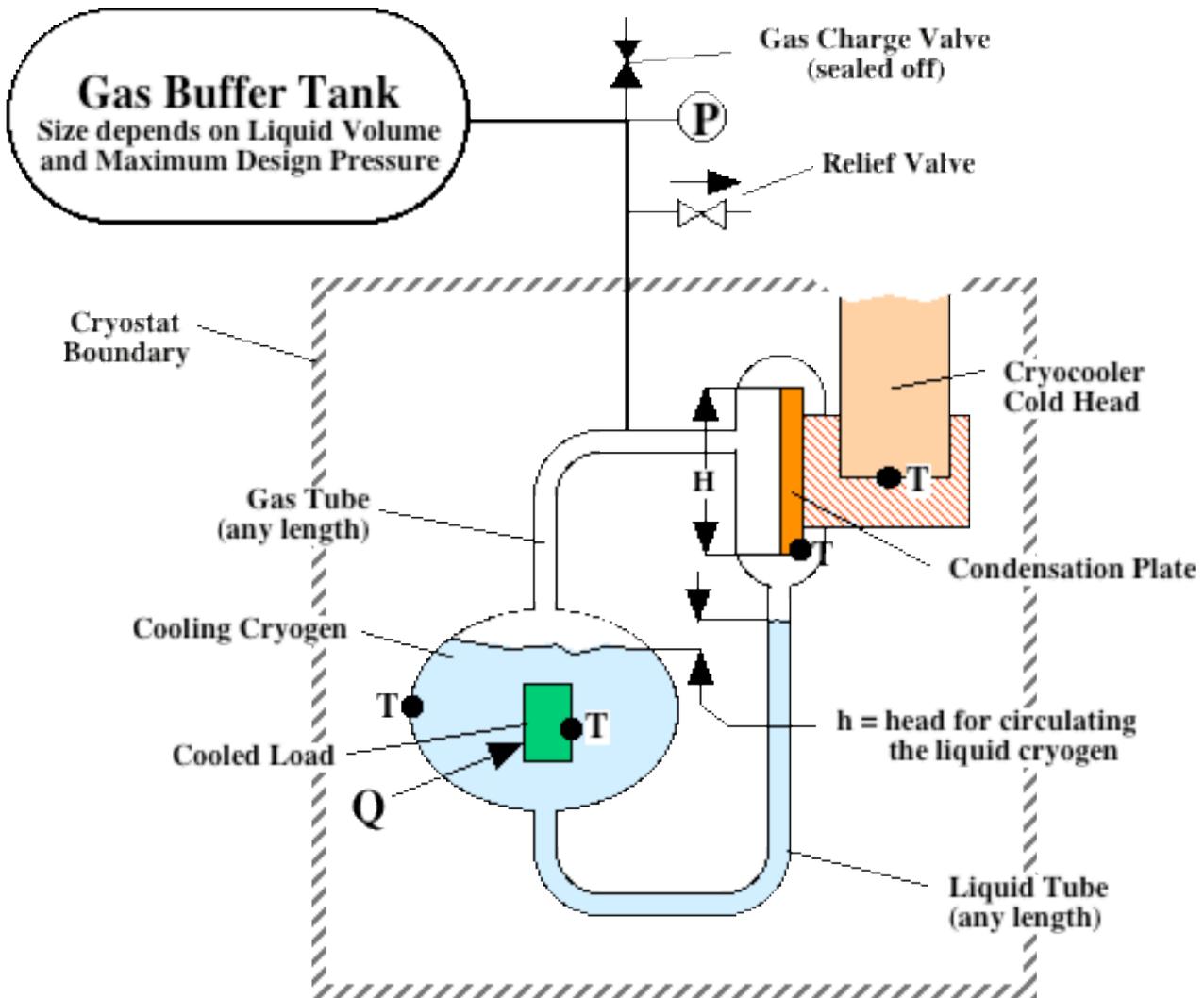


Figure 1. A Schematic Representation of a Heat Pipe Heat Transfer System for an Electronic Device Immersed in a Liquid Cryogen (Note: the system shown is charged with gas at 300 K and left alone.)

Figure 1 shows a schematic representation of a cryocooler with a condenser attached to the cold head. Figure 1 is nothing but a schematic representation of a heat pipe that uses gravity to separate the liquid and gas phases. For the heat pipe to transmit heat from the device being cooled to the cooler cold head, the hotter surface must be in liquid cryogen that is below the condenser surface, which is in gas. The temperature drop ΔT from the load being cooled and the cooler cold head is a function of the heat transfer per unit area Q/A of both the boiling and condensing surfaces, the surface orientation (both surfaces), the height of the condensing surface H , and the fluid properties. In order for a gravity driven heat pipe to function properly, the following must be true: 1) the condensing surface must be oriented so that the fluid flows off the surface in a thin layer; 2) The condensing surface must not be flooded; 3) the boiling surface should face up or to the side (if both sides of a device are cooled); and 4) the pipes between the boiling and condensing surfaces must be large enough to carry the flow with a low pressure drop.

The potential advantages of using a heat pipe to carry heat from the device being cooled to the cooler cold head are as follows: 1) Heat can be transferred from the device to the cold head with a small total temperature drop ΔT . This advantage is particularly useful for devices that generate a lot of heat (say 500 mW) at liquid helium temperature. (The use of a heat pipe in a large device (about 3.1 W) at LBNL reduced the ΔT from the device to the cold head from 450 mK to 30 mK [6].) 2) Coolers that produce temperatures in the 4 K range produce temperature oscillations on the cold head in the range of 0.25 to 0.3 K peak to peak. A heat pipe will filter these temperature oscillations almost completely. The amplitude of cold head temperature oscillations goes down as one goes up in temperature and the negative effects of a given temperature oscillation are smaller for devices that operate at higher temperatures. 3) A heat pipe can carry heat from the device to the cold head over long distances, because the ΔT from the device to the cold head is not a function of the distance between the heat load and the cold head, to first order. This statement applies in all temperature ranges. 4) Because heat can be transferred over long distances almost without penalty, heat pipes can be used to connect multiple devices to a single larger cryogenic cooler. 5) A properly designed heat pipe, (particularly one that carries heat over a long distance) will act as a thermal diode. Heat transfer from the device being cooled to the cold head will be very good. If the cold head is warmer than the device, the heat transfer going from the cold head to the device will be poor. This effect can be dramatic even at liquid nitrogen temperature. The use of a heat pipe as a thermal diode is not a new idea. NASA was doing this over 35 years ago.

The primary disadvantage of using a heat pipe to connect the device to the cooler cold head is the extra complication involved with fabricating the heat pipe system. The pipes must be long and coiled if good vibration isolation is desired. The gravity separation heat pipe shown in Figure 1 will not operate in a zero g environment (there are heat pipes that will) or on devices where the direction of gravity changes more than 30 degrees. The advantages gained from using a heat pipe to transfer heat from the device to the cold head are greatest for devices operating at liquid helium temperature. As one goes up in temperature, the disadvantages of using a heat pipe may outweigh the potential advantages of using a heat pipe to carry the heat from the device to the cooler cold head.

In Figure 1, the electronic device is immersed in a cryogen bath at its saturation temperature T_s . The electronic device boils the cryogen. The cryogen gas goes to the condenser through the tube marked as the gas tube. The liquid drips off the condenser surface and returns to the liquid bath through the tube marked liquid tube. As long as the condenser is physically above the surface of the liquid cryogen (with gravity acting in the downward direction), the cryogen will circulate from the heat source to the condenser as a gas and from the condenser back to the heat source as a liquid. The circulating mass

flow $M = Q/h_{fg}$ where Q is the heat load and h_{fg} is the heat of vaporization of the fluid at its saturation temperature. The room temperature buffer volume shown in Figure 1 permits one to design a completely passive closed loop cooling system that has reasonable gas pressures at room temperature. Since the liquid volumes in a superconducting electronics system tend to be small (less than 50 cc), the volume of a 20 atmosphere buffer volume would also be small (less than 2000 cc).

The heat transfer coefficient for a condenser of height H for a temperature drop across the condenser ΔT_c will take the following general form [7];

$$h_c = 0.94 \left[\frac{\rho_l^2 g h_{fg} k_l^3}{\mu_l H \Delta T_c} \right]^{0.25} \quad -1-$$

where ρ_l is the density of the liquid cryogen; g is the acceleration of gravity; k_l is the thermal conductivity of the saturated liquid; and μ_l is the viscosity of the liquid. H , h_{fg} and ΔT_c are previously defined. From Equation 1 one can define the heat transfer area per unit area in the condenser $(Q/A)_c$;

$$\left(\frac{Q}{A} \right)_c = \left[\frac{\rho_l^2 g h_{fg} k_l^3}{\mu_l H} \right]^{0.25} \Delta T_c^{0.75} \quad -2-$$

The temperature drop across the condenser ΔT_c can be estimated using the following expression;

$$\Delta T_c = \left[\left(\frac{Q}{A} \right)_c \left(\frac{\mu_l H}{\rho_l^2 g h_{fg} k_l^3} \right)^{0.25} \right]^{\frac{4}{3}} \quad -3-$$

From Equation 3, it is clear that the temperature drop on the condensing surface is a function of the heat transfer per unit area on that surface. Extending the surface will reduce ΔT_c provided the fins are vertical so the condensed liquid can flow off of them.

The peak nucleate boiling heat flux per unit area $[Q/A]_{np}$ and the peak temperature difference between the hot surface and the bath ΔT_{np} during nucleate boiling can be estimated using the following expressions [8];

$$\left[\frac{Q}{A} \right]_{np} = 0.16 h_{fg} \rho_v^{0.5} [\sigma g (\rho_l - \rho_v)]^{0.25} \quad -4-$$

$$\Delta T_{np} = 0.5 \left[1 - \frac{T_s}{T_c} \right]^{1.5} T_c \quad -5-$$

where ρ_l is the density of the saturated liquid ; ρ_v is the density of the saturated vapor; σ is the surface tension; T_s is the saturation temperature of the fluid; and T_c is the critical temperature of the fluid. Equation 5 is good over a range of $0.56 < T_s/T_c < 1$. For $Q/A < (Q/A)_{np}$, Q/A is proportional to $\Delta T^{2.5}$.

One would never design a system to operate at the peak nucleate boiling point. It would be prudent to design the boiling system so that boiling occurs at a heat flux per unit area $(Q/A)_b$ well below the minimum film boiling heat flux $(Q/A)_f$. The boiling temperature drop ΔT_b for the boiling side can be expressed as follows;

$$\Delta T_b = \left[\frac{(Q/A)_b}{(Q/A)_{np}} \right]^{0.4} \Delta T_{np} \quad -6-$$

From Equation 6, it is clear that the ΔT_b on the boiling surface is a function of $(Q/A)_b$. If wants to reduce the value of ΔT_b , one must increase the boiling area. Extending the surface by using both sides of the surface and by using fins is useful provided the fins are oriented so that the bubble are not trapped on the boiling surface. As a matter of reference, the minimum film boiling heat flux for various fluids is as follows; for liquid helium $(Q/A)_f = 2000 \text{ W m}^{-2}$; for liquid hydrogen $(Q/A)_f = 7000 \text{ W m}^{-2}$; for liquid neon $(Q/A)_f = 5000 \text{ W m}^{-2}$; and for liquid nitrogen and liquid oxygen $(Q/A)_f = 8000 \text{ W m}^{-2}$ [9]. As with ΔT_c , the value of ΔT_b can be set by controlling the heat transfer per unit area of the boiling surface.

The effective heat transfer coefficient for the combined boiler and condenser can be estimated using the following expression;

$$h_{bc} = \frac{Q}{A_t} \left[\frac{1}{\Delta T_b + \Delta T_c} \right] \quad -7-$$

where Q is the heat generated at the load; A_t is the area of the two tubes; ΔT_b is the boiling temperature drop; and ΔT_c is the condenser temperature drop.

The heat transfer coefficient calculated using Equation 7 should be compared with the effective heat transfer coefficient for a flexible copper strap;

$$h_{Cu} = \frac{k_{Cu}}{L_{Cu}} \quad -8-$$

where h_{Cu} is the heat transfer coefficient for a copper strap with a thermal conductivity k_{Cu} , and L_{Cu} is the length of the strap. The minimum temperature drop between the cryocooler cold head and the load through flexible copper straps (ΔT_{Cu}) can be estimated using the following expression;

$$\Delta T_{Cu} = \frac{Q}{A_{Cu}} \frac{L_{Cu}}{k_{Cu}} \quad -9-$$

The ΔT calculated using Equation 9 does not include temperature drops across solder joints or temperature drops that occur between the load and the end of the copper strap. For example, if the copper strap terminates in a bath of liquid cryogen, there is a temperature drop needed to carry the heat from the load through the cryogen to the strap. Even natural convection heat transfer between the load and the strap will add a lot to the total ΔT between the load and the cryocooler cold head.

When one compares Equations 7 and 8, one sees that the heat transfer coefficient for the condenser boiler system is independent of the distance separating them. When one compares Equations 3 and 6 with Equation 9, one sees that the ΔT between the heat load and the condenser surface (through the heat pipe) is independent of distance, whereas the ΔT along a copper strap is proportional to its length. This means that if the heat pipe system is properly designed, the heat load can be located quite far from the cryocooler and condenser.

Table 2 compares the performance of a well-designed heat pipe with a pure copper strap as a means to transfer heat from an electronic device to the cold head of a cooler. This comparison is done at 4.2 K, 20 K, and 77 K. The fluids used in the heat pipe are helium, hydrogen, and nitrogen respectively. The heat transferred per unit area (Q/A) and the temperature drop ΔT are compared for the two methods over distances of 150 mm and 1000 mm. For the copper strap Q/A is defined as the heat transferred divided by the strap cross-section area. The Q/A used for the heat pipe is based the heat transferred divided by the liquid cross-section area of the liquid tube plus the gas cross-section area of the gas tube. The value of Q/A used for the helium tubes was based on the tubes used for the experiment [6]. The theoretical Q/A for a helium heat pipe may be higher than that used in Table 2. The Q/A for the other fluids is based on scaling the mass flow M for of the fluid and the distance the fluid must travel in a heat pipe. (Note: $M = Q/h_{fg}$, where h_{fg} is the fluid heat of vaporization shown in Table 1.)

Table 2. A Comparison of the Heat Transfer Performance of a Copper Strap and a Heat Pipe

Heat Transfer Method	T (K)	Q/A ($W m^{-2}$)	Length (m)	ΔT (K)
Copper Strap	4.2	2600	0.15	0.48
Copper Strap	4.2	2600	1.00	3.18
Heat Pipe	4.2	25000	0.15	0.09
Heat Pipe	4.2	16800	1.00	0.09
Copper Strap	20.3	2600	0.15	0.18
Copper Strap	20.3	2600	1.00	1.20
Heat Pipe	20.3	89000	0.15	0.18
Heat Pipe	20.3	60000	1.00	0.18
Copper Strap	77	2600	0.15	0.49
Copper Strap	77	2600	1.00	3.25
Heat Pipe	77	64000	0.15	0.47
Heat Pipe	77	43100	1.00	0.47

The calculation for the heat pipe are based on the following values heat transfer per unit area for the condensing surface and the boiling surface: for helium, $(Q/A)_c = 60 W m^{-2}$ and $(Q/A)_b = 10 W m^{-2}$; for hydrogen, $(Q/A)_c = 240 W m^{-2}$ and $(Q/A)_b = 40 W m^{-2}$; and for nitrogen, $(Q/A)_c = 360 W m^{-2}$ and $(Q/A)_b = 60 W m^{-2}$. The residual resistance ratio RRR for the copper strap is assumed to be 130. If the strap RRR is higher, the ΔT will be lower at 4 K and 20 K. On the other hand, if the copper is cold worked so that the RRR goes down, the ΔT at 4 K and 20 K will be higher than shown in Table 2. (The RRR of the copper has a relatively small effect on strap heat transfer performance at 77 K.)

Based on Table 2 and experimental evidence [6], the use of heat pipe can improve the heat transfer from the device to the cold head at all temperatures. The effective heat transfer coefficient is higher for the heat pipe for all of the cases shown in Table 2. (See Equations 7 and 8.) The heat pipe improvement over a copper strap is particularly pronounced at liquid helium temperatures. If one is going to carry over 0.5 W of heat from a load or a group of loads, use a heat pipe, if these loads are at 4.2 K. When a device has low heat load (say 10 mW), one can reduce the Q/A for the copper strap so that the ΔT is acceptable, but one may still have to deal with the temperature oscillations in the cold head. If it is desirable to transfer a larger amount of heat over long distances, the heat pipe is a clear winner.

When cooling devices at 20 K and 77 K, the case in favor of a heat pipe is less appealing. There is improved heat transfer performance for the heat pipe over distance as compared to a copper strap. At all temperatures, the cross-section area of the elements (straps or tube) is much smaller for a given heat flow when one uses a heat pipe. Heat pipes by their very nature require more space and longer distances. Heat pipes require the heat load to be in a cryogen bath and a heat transfer element based on a heat pipe is more complicated. The choice of using a heat pipe or a copper strap to carry heat from the load depends on the details of the system being cooled and the temperature requirements of the device that is being cooled.

VI. MONITORING AND CONTROL OF THE REFRIGERATION DELIVERED FROM A CRYOCOOLER

A common method for controlling and monitoring cooling performance is to monitor the liquid level in a cryostat filled with liquid cryogen. This may be the wrong thing to do, particularly in a narrow cryostat neck, when a cryogenic system is cooled using a cryocooler. When the temperature drop between the load and the cryocooler is large, the cryostat may be warming while the temperature of the cold head of the cryocooler is much colder than the load.

The expansivity of the liquid cryogen used as a medium to transfer heat from the device to the cryocooler cold head may be an issue. (This is particularly true for systems using liquid helium as a coolant.) Using the level of the liquid cryogen to monitor cooling performance can be the wrong thing to do, because the liquid level in a device can rise as the cryogen gets warmer due to the expansivity of the liquid. This lesson was recently learned the hard way at LBNL.

Cryogen liquid level in cryostat should not be used to monitor or control the cooling process. The temperature of the cryocooler cold head and the temperature of the load (and the pressure in the liquid container if the load is surrounded by liquid cryogen) should be used to monitor the performance of the cooling system. Suggested locations for temperature sensors (a black dot with a T next to it) in a heat pipe cooling system is shown in Figure 1. A large temperature difference between the load (or the fluid around the load) and the cryocooler cold head is often a symptom of bad cryogenic system design or a fault that is interfering with the heat transfer between the load and the cold head. It is good practice to measure the temperature of the cryocooler cold head and the load.

Measurement of the liquid cryogen temperature and the condenser temperature are good intermediate temperatures to measure. The cryocooler cold head temperature is a reliable measure of cryocooler performance. For a number of cryocoolers that have been studied, the temperature of the cold head is a linear function of the heat entering the cold head. The difference between the temperature of the load and the cryocooler cold head can also be a measure of the heat being produced by load. If both the cold head temperature and the temperature difference between the load and the cold head go up, the heat load is increasing. If the cold head temperature goes up but the temperature difference between the cold head and the heat load is unchanged, the cryocooler performance may be degraded. An increasing cold head temperature may be a sign that the cryocooler needs maintenance or that there are additional heat leaks into the cryocooler cold head. An increase in the temperature difference between to load and the cold head with no change in the cold temperature indicates that there is a degradation of the heat transfer coefficient between the load and the cold head. In a system that uses one or more heat pipes to carry the heat from the load to the cold head, this may be an indication of partial flooding of the condenser surface with liquid. In liquid helium systems, extensive flooding often results in a run away rise of the load temperature while the cold head temperature goes down.

VII. CONCLUDING COMMENTS

Experimentalists will continue to cool superconducting electronic devices with either liquid helium or liquid nitrogen. The use of stored liquid cryogens may remain viable for some sorts of devices when they are used in the field. As electronic devices become larger, there is desire to use mechanical coolers for cooling the devices. In general, the most reliable cryogenic coolers are closed cycle devices that produce their cold on the surface of a cold head. There is a continuing effort to develop more reliable and more efficient closed cycle mechanical coolers that can be tied to electronic devices.

The problem of transmitting the heat from the device to the cold head become more pronounced in larger systems. Vibration is a serious issue for some types of electronic devices, so these devices can not be mounted directly on the cold head. Vibration isolation can be accomplished using flexible copper straps or by the use of a heat pipe. When the heat loads are low and the device is operating at temperatures above 20 K, the use of a copper strap is a perfectly adequate solution to the vibration isolation problem. For devices that operate at temperatures below 20 K, oscillations of the cold head temperature become another performance limiting factor. Heat pipes can filter out the temperature oscillation present on the cold heads of low temperature coolers while reducing the temperature drop between the load and the cold head. Heat pipes also permit the cooler to be moved some distance from the load (in all temperature ranges). The only way to reliably monitor the performance of a cooling system for electronic devices is to measure the temperature of the cold head and the load.

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REFERENCES

- [1] V. J. Johnson Editor, A Compendium of Materials at Low Temperature (Phase 1), Pergamon Press, New York (1961)
- [2] M. A. Green, "The Effect of Low Temperature Cryocoolers on the Development of Low Temperature Superconducting Magnets," IEEE Transactions on Applied Superconductivity **11**, p 2615, (2001)
- [3] D. S. Buchanan, D. Paulson, and S. J. Williamson, "Instrumentation for Clinical Applications of Neuromagnetism," Advances in Cryogenic Engineering **33**, p. 97, Plenum Press, New York, (1987)
- [4] G. N. Ellison, E., L. Hershberg, and D. L. Patalzick, "The Cryocooler Implications of Flexible, Multi-channel I/O Cables for Low Temperature Superconducting Microelectronics, Advances in Cryogenic Engineering **39**, p 1169, Plenum Press, New York, (1993)
- [5] C. E. Taylor, "Design of a Helium Re-condenser for the Venus Cryostat," LBNL Superconducting Magnet Group Report 759, January 2002
- [6] C. E. Taylor, C. M. Lyneis, D. Leitner, and S. R. Abbott, "An Efficient Cooling Loop for Connecting a Cryocooler to a Helium Reservoir," to be published in Advances in Cryogenic Engineering **49**, AIP Conference Proceedings, (2003)
- [7] F. Kreith, Principles of Heat Transfer, International Textbook Company, New York, (1961)
- [8] T. K. H. Frederking, "Peak Heat Flux and Temperature Difference in Nucleate Boiling of Liquefied Gases," Advance in Cryogenic Engineering **8**, p 489, Plenum Press, New York, (1962)
- [9] E. G. Brentari and R. V. Smith, "Nucleate and Film Pool Boiling Design Correlations for O₂, N₂, H₂, and He," Advance in Cryogenic Engineering **10**, p 325, Plenum Press, New York, (1964)

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